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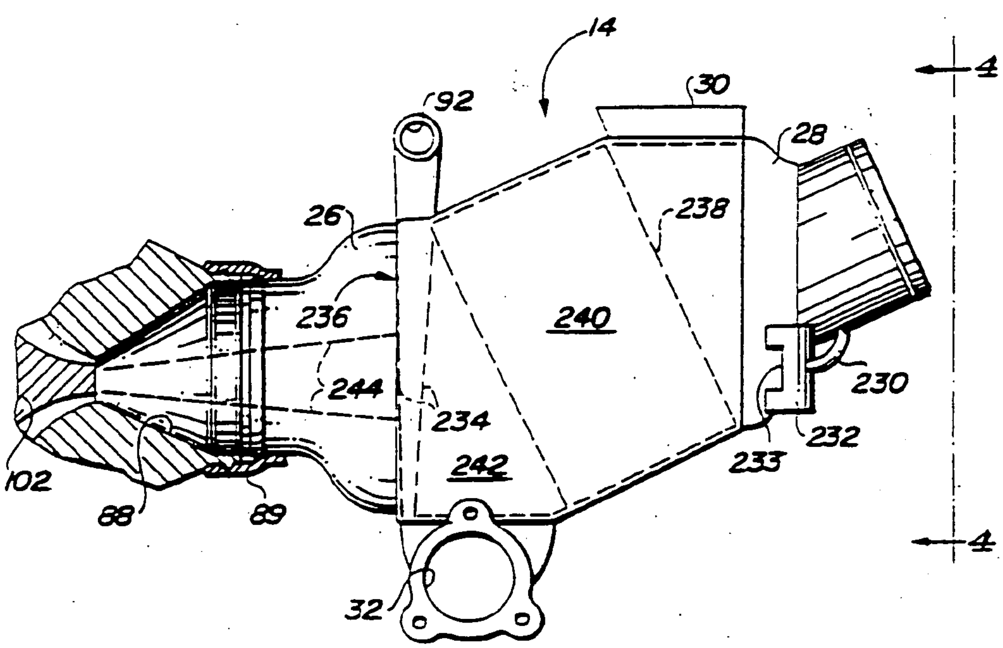

(57) Abstract

A system for conditioning working fluid in environmental control systems (10, 310) includes arrangements for minimizing icing from a variable flow velocity turbine exit flow (88) at subfreezing conditions, wherein the turbine (22) is very closely located to the downstream heat exchanger (14), including a backpressure plate (246) for minimizing flow velocity stratification, an internal bypass passage (254) arranged to produce a relatively predictable bypass flow ratio regardless of the flow velocity stratification, and other anti-icing techniques (232, 260, 346, 454).

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FLUID CONDITIONING APPARATUS AND SYSTEM

BACKGROUND OF THE INVENTION

1. Field Of The Invention

This invention pertains to the art of fluid conditioning apparatus, systems and methods, and is more particularly concerned with improvements for preventing excessive ice formation at the cold fluid inlet of heat exchangers in these systems, as well as for providing improved heating capacity utilizing the heat transfer performance capabilities of heat exchangers.

2. Description Of Prior Art

Examples of arrangements for preventing excessive ice formation are described in three patents. The first is U.S. Patent No. 4,198,830 dated April 20, 1980, of Carl D. Campbell, and entitled "Fluid Conditioning System And Apparatus". The second is U.S. Patent No. 4,246,963 dated January 27, 1981, of Alexander Anderson, and entitled "Heat Exchanger". The third is U.S. Patent No. 4,352,273 dated October 5, 1982, of Robert C. Kinsell et al, and entitled "Fluid Conditioning Apparatus And System". All of the above patents are assigned to the same assignee as that of the present invention.

Briefly stated, the Campbell invention provides a means to condense water out of the working fluid of an air-cycle environmental control system while the fluid (air) was still at high pressure, eliminating the need for coalescer bags which r quire significant maintenance. A problem with ice formati n on the fac of th heat

exchanger in line with the discharge of the expansion turbine was addressed by the Anderson invention which proposed the addition of hot header bars, not connected to the cold passages, for improved ice prevention characteristics of the heat exchanger. The Anderson invention also detailed the heat transfer relationship between the hot and the cold side fins necessary to maintain metal temperatures above freezing. The Kinsell et al invention provides for the substantial lowering of the expansion turbine discharge air temperature and an associated increase in the delivered cooling capacity of a typical apparatus by adding a bypass in the middle of the heat exchanger.

In addition to the type of air cycle environmental control system (ECS) described in those patents, the present invention is useful in an ECS system variation wherein condensing heat exchanger is used to achieve extra heating capacity in a combined ECS, nuclear-biological contaminant filtration system while maintaining the proper environmental conditions at the filter inlet. This cycle also relies on the three previously described patents for the prevention of ice in subfreezing condensers. The above inventions have found successful application in both commercial and military air and ground vehicle environmental control systems. However, the methods described above for controlling the formation of ice have normally required additional, active methods of ice control.

Applications of the Anderson invention have shown that the performance of the condenser heat exchanger changes dramatically with changes in operating conditions, specifically turbine discharge velocities. This is due to the design of the heat exchanger cores described in the

Anders n invention. The variation in performance is due to the very low pressure loss of the condenser core relative to the manifold pressure losses. In other heat exchangers, core pressure losses are typically 80% f th flange-to-flange pressure loss to ensure proper flow distribution through the heat exchanger. The Anderson invention provides for very loose fins on the cold side passage of the heat exchanger to prevent the accumulation of ice by reducing the blockage of the flow path and by biasing the metal temperatures closer to the hot side than the cold side. The resulting design is therefore very sensitive to flow stratification at the condenser cold side inlet.

Testing of various systems utilizing the referenced inventions has shown that, when the turbine is close-coupled to the condenser, as it is in most systems, changes in turbine exit velocity produce significant changes in the flow and temperature stratification f the condenser cold side inlet.

SUMMARY OF THE INVENTION

An important aspect of the present invention is a postulation that this above-described stratification produces significant deviations from predictions of metal temperatures for these prior arrangements. The concentration of cold air from the turbine exhaust in the center of the condenser core results in lower than predicted metal temperatures locally, and the formation of ice when no ice is predicted. This is believed to be a primary reason for use of secondary, active de-icing devic s in th se systems. Additionally, the amount f heat transfer produced by the h at exchanger varies significantly from predictions in which stratification are

not accounted f r, resulting in oversized condensers to meet water removal or heating requirements. Also, there is a problem with the predictability of heat exchanger performance in off-design conditions where turbine exit velocities vary significantly from design cases.

With these problems in mind, the present invention is intended to eliminate the variation in condenser heat transfer performance while retaining the benefits of the Anderson invention for ice protection. This will allow the Campbell, Anderson and Kinsell inventions to function properly in typical aircraft installations where the condenser heat exchanger of an air-cycle system is close-coupled to the expansion turbine. This will also allow repeatability and predictability in performance predictions in off-design conditions.

More particularly, the present invention recognizes that the condenser heat exchanger described above, in order to perform consistently and avoid excessive ice formation in conditions of inlet stratification experienced when close-coupled to the turbine exit, must be designed in such a manner that the inlet air is properly distributed to the cold side inlet of the heat exchanger.

The apparatus and method of the present invention for distributing the inlet flow, comprises a backpressure plate attached to the back of the condenser heat exchanger core, which is offset from the core to allow flow through all passages. The design of this plate is such that the overall pressure loss of the condenser is higher than without the plate, allowing the r tention of the hot and c ld fin heat transfer relationship described in the Anderson invention. By placing th plate on th back side

of the c re, the turbine exit air has been reheated in th condenser heat exchanger and therefore the plate will not collect ice. The resultant effective flow area of the core is less than the inlet manifolds, and this results in a more normal relationship between core and manifold pressure losses.

Another important aspect of the present invention is the recognition that such flow stratification may have a significant impact in the operational aspects of an integral bypass duct or gap configuration as taught in the Kinsell et al patent. Testing has shown that, when the turbine is close-coupled to the condenser, as it is in most systems, changes in turbine exit velocity produce significant changes in the flow and temperature stratification of the condenser cold side inlet. This stratification produces significant variation in the percentage of flow passing through the gap.

With this problem in mind, this invention is intended to eliminate the variation in condenser bypass ratio while retaining the benefits of the Kinsell invention for ice protection. This will allow functioning, repeatability, and predictability in performance predictions in the systems. More specifically, the invention recognizes that the condenser gap described above, in order to perform consistently in conditions of inlet stratification experienced when close-coupled to the turbine exit, must be placed in such a manner that the gap is outside of the turbine exit high velocity area, so that changes in the turbine exit velocity will have minimal impact on the designed condenser bypass ratio. Testing has shown that an internal location f the gap on the extreme side of the condenser, rather than in the middle, accomplishes this purp se. In order t keep the outside wall of the gap

from freezing, a small, one-pass heat exchanger is required to keep the metal temperatures sufficiently warm.

Another aspect of the present invention is improved bypass duct capacity while avoiding icing therein. Specifically, inclusion of a closure plate over a part f the bypass duct, near its exit, but in a manner avoiding reduction in size of the smallest dimension of the duct, allows lower bypass ratios without increasing ice formation. Additionally, the present invention contemplates anti-icing by the convenient utilization of hot, waste fluid flow from the system, such as exhaust flow from fluid film foil bearings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic representation of a fluid conditioning system embodying the principles of the present invention;

FIG. 2 is a partial, cross-sectional elevational view of the air cycle machine 20;

FIG. 3 is a partially schematic, partial fr nt elevational view of the condenser heat exchanger 14;

FIG. 4 is a right side elevational view, taken along lines 4-4 of FIG. 3;

FIG. 5 is an enlarged cross-sectional, front elevational view of the core portion of heat exchanger 14 with the fluid plenum casings thereof not shown, as viewed generally along lines 5-5 of FIG. 6;

FIG. 6 is a right side elevational view of the outlet portion of the heat exchanger core as viewed al ng line 6-6 of FIG. 5;

FIG. 7 is a left side elevational view of the inlet portion of the heat exchanger core as viewed along line 7-7 of FIG. 5;

FIG. 8 is a schematic depiction of flow through bypass duct 254;

FIG. 9 is a view similar to FIG. 1 but showing an alternate fluid conditioning system embodying an alternate heat exchanger 314;

FIG. 10 is a schematic depiction of the heat exchanger 314; and

FIG. 11 is a view similar to FIG. 10 but showing another embodiment of heat exchanger 414.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

One form of the apparatus of the invention, and one form of the system employing the apparatus, is illustrated on FIG. 1 wherein the system 10 for the conditioning of the air supplied from a source to an inlet 11, a point of use as, for example, the cabin of the aircraft, through an outlet 13, employs an apparatus 12 comprised of a first heat exchange means 14, a second heat exchange means 16, a water trap or separator 18 and an air cycle machine 20. The machine 20 is of the three-wheel type generally known in the art, comprising an air expansion turbine 22 and a centrifugal type air compressor 24 with the turbine and

the compressor wheels mounted on a common rotatable shaft, together with a fan wheel.

The heat exchange means 14 in its preferred form is of the type familiarly known in the art as a counterflow plate-fin device having fin means (not shown on FIG. 1) defining a first fluid passageway means separated by plate means (also not shown on FIG. 1) from a second passageway means defined by counterflow-disposed fins (also not shown). Fluid flowing from a chamber defined by th plenum casing 26 through the first passageway means enters a chamber defined by a plenum casing 28 after having been in heat exchange relationship with fluid flowing through the second passageway means from a chamber defined by the plenum casing 30 and entering a chamber formed by th plenum casing 32. The counterflow plate-fin-device 14 is described in greater detail hereinbelow with respect to FIGS. 3-8.

Reference may be made to the Kinsell U.S. Patent 4,352,273 for a detailed description of the majority of elements in FIG. 1, including their operation and interrelationship. This would include the second heat exchange means 16 with its chamber forming plenum casings 34, 36, 38, 40, third heat exchange means 42 with its chamber forming plenum casings 44, 46, fourth heat exchange means 48 with its chamber forming plenum casings 50,52, coolant flow duct 54 with admitting and discharging ends 56, 58, fan wheel 70, water pipe 51 and supply nozzle 53, and ducts 74, 76, 78, 80, 82, 83, 88 and 90.

Additionally, for temperature regulation, a bypass passage 60 extends from duct 74 to duct 76 in parallel to the primary h at exchanger 48, and mixing valves 62,64 may

b variably adjusted to control the relative airfl w rates through the two parallel paths and thereby adjust temperature of airflow reaching compressor 24. Similarly, a bypass passage 66 extends from duct 78 to duct 80 in parallel to the secondary heat exchanger 42, and mixing valves 68,72 may be variably adjusted to control the relative airflow rates through these two parallel paths to control the temperature of airflow reaching plenum casing 38. As shown, a single actuator 71 may be used to simultaneously further open one and close the other of valves 68,72. A parallel bypass passage 85 extends from duct 84 in non-heat exchange, non-heated relationship to reheater 16, to control the temperature of airflow delivered to downstream duct 86. As before with respect to bypasses 60 and 66, a pair of valves 87,89 variably control airflow to plenum casing 34 and bypass 85 (as by a single actuator as illustrated if desired) for temperature control.

Interposed in duct 86 is a filter means 91 for extraction of certain contaminants from the airflow prior to its introduction into habitable airspace. An example of such a filter would be a charcoal type filter for absorption of biological or nuclear contaminants. Such type filters have a life and efficiency highly sensitive to temperature and humidity, requiring in the environmental control system of the type illustrated in FIG. 1, that the filter be located downstream of reheater 16. Even so, in certain operating conditions the temperature of airflow exiting plenum 36 of the reheater could become excessive, requiring use of bypass 85 to limit temperature of airflow in duct 86. Additional eff cts of the necessarily limited temperature environs for filter 91 ar discussed in gr at r detail below.

Upon exiting filter 91, a small portion of the airflow may be diverted from duct 86 to a secondary outlet 93. As well known to those skilled in the art, outlet 93 may extend to an oxygen-nitrogen generator (not shown) which operates through reverse osmosis processes to create oxygen-enriched and nitrogen-enriched airflows for other uses within the aircraft environment.

From filter 91 the relatively hot airflow passes t the inlet of the expansion turbine 22 where it is expanded and cooled to a point consistent with the energy imparted by the turbine 22 to the compressor 24 and the fan wheel 70. As will be apparent to those skilled in the art, the turbine and compressor operate in what is familiarly known as boot strap fashion.

In accord with U.S. Patent 4,246,963, condenser 14 is of the type having hollowed, heated header bars traversing the cold air inlet adjacent plenum 26. A duct 92 therefore extends from duct 80 to these header bars (not shown in FIG. 1) to pass a portion of hot air therethrough to assist in preventing excessive ice formation. As known, the de-icing flow from duct 92 and the header bars is discharged into plenum 32 to mix with the cooler air therein flown from the second passageway means.

The system illustrated in FIG. 1 is of the type wherein all airflow to pass into the habitable space from outlet 13 is routed through the above-described flow path in FIG. 1. Thus, for example, all airflow is thereby treated by filter 91. Heat exchanger 14 is therefore necessarily sized to accomplish two purposes: condensation of th m isture in the hot airflow passing from plenum casing 30 to plenum casing 32; and also to acc mplish significant heating of the subfreezing airflow

that enters plenum casing 26 so as to produce the desired temperature at outlet 13. This heating function requires a substantially greater size (e.g., three-times size) heat exchanger 14 than required merely for the condensation function. Because of this relatively large heat exchang size, the usual passive manners of preventing ice formation within heat exchanger 14 (e.g., those taught by Anderson U.S. Patent No. 4,246,963 and Kinsell U.S. Patent No. 4,352,273) do not produce fully acceptable results throughout the operating envelope of the system of FIG. 1. At the same time overall system efficiency is maintained by essentially always producing subfreezing outlet flow in duct 88. Further, tendency toward icing in heat exchanger 14 is also increased because of the close coupling of turbine 22 to plenum casing 26 to reduce the length of duct 88 to minimize icing in the latter (and accordingly minimize de-icing techniques for duct 88 itself).

It is to be understood that, as used herein, the description of the FIG. 1 system as being one in which "all" fluid flow passes through filter 91, refers to the FIG. 1 system in its filter-operating functional mode and the associated heating function for heat exchanger 14. Specifically, it is within the scope of the invention and the terms defined in this patent, that a selective bypass of filter 91 may be included to modulate the duty cycle thereof to increase its life. Such, of course, does n t alter the capacity requirement of heat exchanger 14.

While heat exchanger 14 is discussed in gr ater detail below, one aspect of passive anti-icing control is accomplished by delivery of an otherwise waste-heat airflow through a passage 230 extending from the air cycle cooling machine 20 to a warming chamber 232 on casing 28.

To more fully explain the source of the heating flow in passage 230, air cycle cooling machine 20 is illustrated in greater detail in FIG. 2. Reference may be made to FIG 4 of U.S. Patent 4,507,939 of Wieland for a detailed description of the majority construction components in FIG. 2 including their operation and interrelationship (with the exception that the Wieland patent further includes a central passage and third center wheel fan). This would include housing 98 with its outer and inner portions 98a, 98b and inlet passage 100, turbine wh el 102, housing 104 with outlet 106, discharge passage 108, and annular diffuser section 110, passage 112 associated with compressor 114 inner and outer shafts 116, 118. Also described in greater detail therein are gas foil thrust bearing 174 and a gas foil journal bearing 176, similar in construction to those shown in U.S. Patent 3,615,121, and their gas lubrication flow path including elements 216, 22, 224 and ultimately discharge 230, for developing pressurized gas film supports in clearance spaces such as 196, 206 and 212.

In addition, the airflow in clearance space 212 functioning with the gas foil bearing thereof experiences substantial heating prior to exhausting from the bearing and the housing 140 to passageway 230. This relatively hot, waste airflow is directed to the heat exchanger 14 as described above with respect to FIG. 1 for anti-icing purposes.

Returning now again to the heat exchange means 14, this heat exchanger is shown in greater detail in FIGS. 3-7. As best shown in FIG. 3, the exit duct 88 from the turbine 102 is very closely coupl d to the inlet plenum casing 26 of heat exchanger 14. A sealed connection between the adjacent ends of duct 88 and plenum casing 26 may be pr vided by a sealing and closure element 89. The

hollowed heated header bars are schematically illustrated in FIG. 3 by the dashed lines 234, and these header bars, supplied by heated air from duct 92, are disposed adjacent the front face 236 of the heat exchanger core itself. The counterflow core of the heat exchanger is depict d by dashed lines in FIG. 3 with the flow from inlet plenum casing 26 passing from left to right in FIG. 3 to exit through plenum casing 28. The heated airflow from plenum casing 30 passes downwardly, with respect to FIG. 3, into the heat exchanger core to be turned in the triangular section 238 to then flow in direct counterflow relation to the other cool flow in the rectangular section 240, before then again being turned downwardly in the triangular section 242 to exhaust through the plenum casing 32. As such, the internal core configuration of the counterflow heat exchanger 14 is of conventional construction well-known to those skilled in the art. These triangular and rectangular sections 238, 240, and 242 are depicted by the associated dashed lines more completely in FIG. 5. The general direction of the cool flow passing from plenum 26 to plenum 28 is noted by the solid arrows in FIG. 5, while the relatively warm counterflowing air passing from plenum 30 to plenum 32 is depicted by the dashed arrows in FIG. 5 for clarity.

As can be most clearly seen with respect to FIGS. 3 and 7, the relative close-coupling of the turbine 102 t the front face 236 of the heat exchange core has been found to cause a significant distribution of airflow velocities across the inlet face 236. As depicted by the conical dashed lines 244 in FIG. 3 and their associ ted circular projection as depicted in FIG. 7, a circular central portion of the face 236 receives the cold airflow from turbine 102 at highest velocities. Effectively the airflow in pl num casing 26 would b distributed somewhat

in a Gaussian statistical distributional pattern acr ss the face of inlet 236 with respect to the velocity thereof. In other words, the exhaust plume of airflow from turbine 120 produces a central area depicted by dashed lines 244 of high airflow velocity. More particularly, the characteristics of diffusion of the turbine exhaust would be similar to entrainment of a j t discharge into a plenum, where velocity and flow diffuses into the volume in the profile of Gaussian distribution curve. Accordingly the zone within the circular dashed lines 244 represent the bulk of the inlet flow (or the center of the Gaussian curve) and the highest velocity of airflow. Such distribution of the airflow presumes substantially zero or very low flow restriction through heat exchanger 14 with respect to the cold subfreezing airflow passing from plenum 26 to plenum 28. As a result, this same stratification of airflow would extend thr ugh the heater core.

It is quite clear therefore, by observation of FIG. 7, that a smaller circle 244 would result in less heat transfer, since a larger portion of the heat exchange core would be relatively starved of airflow. Conversely, a larger circle 244 would result in greater heat transfer as more and more airflow is spread across a greater percentage of the core. This distributional stratification, i.e. the circular pattern 244 is directly related, of course, to turbine exit velocity.

Turbine exit velocity can vary significantly throughout the duty cycle of the entire environmental control system. Testing has shown that when the turbine 102 is close-coupled to the heat exchanger 14, that changes in turbine exit velocity airflow pr duce significant changes in the flow and temperature

stratification at the inlet face 236. This stratificati n produces significant deviations from expected predictions of metal temperatures. Specifically, the concentration of relatively cold air from the turbine exhaust within the zone bounded by dashed lines 244 results in lower than predicted metal temperatures at these locations. This therefore may create ice formation where otherwise ice would not be predicted. Additionally, the amount of heat transfer between the two counterflowing airflows varies significantly from predictions wherein such stratificati n exists. Typically this has been addressed in the pri r art by a substantial increase or oversizing of the heat exchanger 14 to meet both water removal and heating requirements. Additionally, the predictability of the heat exchanger 14 performance at off- design conditions is extremely problematic where turbine exit velocities vary and are stratified as discussed above.

Accordingly, an important aspect of the present invention is the minimization of stratification of turbine velocities at inlet face 236 to avoid excessive ice formation. At the same time, since the turbine discharge flow rate may vary substantially in the FIG. 1 system t accommodate the various modes of operation thereof, the invention allows operation of the system by reducing ice formation in varying conditions of turbine discharge flow velocities. That is, as used herein, minimization of the flow velocity stratification encompasses actual reduction thereof, and/or increased variations in permitted turbine discharge velocities. One manner in which the present invention contemplates such a result is the inclusion of a backpressure plate 246, shown in FIGS. 4-6, disposed within the outlet plenum casing 28. More particularly, backpressure plate 28 is a relatively thin, rigid, flat metal plate secured adjacent t , but spaced slightly away

from the back side face 248 of the heat exchange core. Backpressure plate 246 is offset a sufficient distance from face 248 to allow substantially unobstructed flow through all of the heat exchange core passages. As illustrated, it may be straightforwardly rigidly secured to back face 248 through securing elements 250.

The design, configuration and location of backpressure plate 246 is such that the overall pressure loss of the airflow passing from plenum casing 26 to plenum casing 28 is higher than would be experienced in the absence of plate 246, thereby allowing an increase in retention of the hot and cold airflows in heat transfer relationship inside the core of the heat exchanger, as well as increasing the retention time of the cold subfreezing air adjacent the hot header passages 234. Placement of plate 246 on the back side of the h at exchanger core assures that the airflow impacting the inside surface 252 of plate 246 has experienced substantial warming so that plate 246 itself will not tend to collect ice. Inclusion of backpressure plate 246 covering a substantial portion of the rear face 248 has been observed to produce an improved airflow distributi n (i.e., reduction of the stratification as depicted by the dashed circle 244) so that more correct predictions of heat transfer performance and metal temperatures within the interior core of the heat exchanger 14 can be predicted. Three factors considered together determine the amount of blockage required (that is the size required of backpressure plate 246) to accomplish the desired goals of increased airflow retention time and reduced flow velocity stratification across the front face thereof. Sp cifically, these three factors include th distance between turbine 102 and c re face 236, the pressure drop experienced acr ss the c re f the airflow passing from

face 236 to face 248, and the turbine exit velocity. It is believed plate 254 would be effective in other specific systems if it covered at least 30 percent of face 248, or between about 30 percent and 80 percent of that face.

In a particular system, for example, backpressure plate 246 covering approximately sixty percent of the face 248 has resulted in sufficient blockage of the airflow to reduce the velocity stratification across face 236 to avoid localized initiation of icing thereon, improved heat exchange flow, and resulting overall acceptable operati n of the system of FIG. 1. In that particular instance, the spacing between surfaces 248 and 252 was approximately five percent of the overall length of the heat exchange core extending from surface 236 to 248. As will be apparent to those skilled in the art, spacing between faces 248 and 252 is important to consideration of efficiency of the overall system sends a major comp nent of pressure loss of the exiting airflow is in the turning loss of the air after it exits surface 248 and flows around plate 246, along with a second turning loss as this airflow rejoins the air exiting plenum casing 28.

The present invention further contemplates a modification and adaptation of the heat exchange apparatus as disclosed in Kinsell U.S. Patent No. 4,352,273. More particularly, the Kinsell et al patent discloses a heat exchanger having a bypass passage integrally constructed to be contained within the same heat exchange enclosure as the heater core carrying the primary flow. Such bypass duct of the Kinsell et al arrangement was found to dramatically improve the overall efficiency of the heat exchange arrangement by all wing lower subfreezing temperatures at the outlet for purposes of cooling, while at the same time anti-icing conditions were also impr ved

inasmuch as the tendency toward icing in the primary passage increased the pressure drop therethrough to permit more air through the bypass. This therefore reduced the cold air traversing the heat exchange core while the same amount of warm air in the counterflowing passage ways was available. This increased the temperature of the primary airflow thereby reducing tendency towards initiation of icing. To the extent necessary reference may be made to the Kinsell et al patent for a more clear understanding thereof, and to the extent required the Kinsell et al patent is incorporated herein by reference.

The arrangement disclosed in the Kinsell et al patent is not directly applicable to the arrangement illustrated in FIGS. 1-7 because the turbine exit velocities may vary significantly when passing through its duty cycle f providing either cooling or heating exhaust airfl w through outlet 13. In particular, during operational conditions of the environmental control system illustrated in FIG. 1 wherein there exists the greatest variation in turbine exit velocity from the standard design condition for heat exchanger 14, the percentage of bypass flow as would be introduced by the Kinsell et al concept, is m st critical. This condition occurs during altitude heating condition when fan 70 is unloaded by ram air pressure in duct 54 and the lower density of air flow therethrough due to altitude, all combined with the fact that the backpressure on turbine 102 is also lowest at the same time. This is particularly true with respect to partially pressurized or unpressurized aircraft. In these conditions the air cycle machine 20 still operates at its highest normal speeds and therefore produces the highest turbine exit velocities.

The pr sent invention solves these problems by structure to minimize variation in the bypass ratio of airflow passing from plenum casing 26 to plenum casing 28, while still retaining the anti-icing benefits as disclosed in the Kinsell et al patent. More particularly, this aspect of the present invention is accomplished by including a gap or bypass duct 254 in the core of heat exchanger 14 equivalent in function and operation to the gap or bypass duct disclosed in the Kinsell et al patent. Gap 254 is disposed at one extreme side of the heat exchanger core instead of being located generally in th center of the heat exchanger core as disclosed in the Kinsell et al patent. In this manner the present invention recognizes that the bypass duct or gap 254, in order to perform consistently in conditions of turbin inlet velocity stratifications caused by the close coupling of turbine 102 to inlet face 236, must be located so that the bypass duct is preferably outside the location of high velocity flow as experienced within the area circumscribed by the dashed line circle 244 in FIG. 7. By placement of bypass duct 254 to the extreme side of inlet face 236, the airflow velocity at this side location d es not vary substantially even though turbine exhaust velocities may be varying dramatically. As a result the ratio of bypass flow passing through bypass duct 254, in comparison to the remainder of airflow passing through the heat exchange core in heat exchange relationship with the warming airflow, can remain predictable.

At the same time, to avoid freezing and additional icing conditions, a small one pass counterflow heat exchange passage 256 is contained between the bypass duct 54 and the outside sid wall 258 of the entire heat exchanger 14. The one pass passage 256 carries count rflowing warm air flow which is passing from plenum

casing 30 to plenum casing 32, rather than carrying the cool flow passing from plenum casing 26 towards plenum casing 28.

Additionally, placement of bypass duct 254 immediately adjacent one sidewall of the heat exchange core and in non-alignment with the central inlet of the turbine airflow passing into inlet plenum 26, has been found to provide additional benefits. Specifically, high velocity ice particles which are normally entrained in the turbine exit velocity airflow will not normally reach the bypass duct 254. Such ice particles being more dense than the air carrying them, will tend to pass through the center portion of the heater core and will stay away from the bypass duct 254. This avoids tendency towards icing across bypass duct 254 which can bridge the gap thereof once ice tends to start accumulating within bypass duct 254.

In addition to improved predictability of heat exchanger operation because of the relative constancy of the bypass ratio because the airflow entering bypass duct 254 is far more predictable because it is outside the high velocity area 244, this aspect further improves the operation of the FIG. 1 arrangement by permitting a smaller size heat exchanger core in heat exchange means 14 due to the reduced bypass ratio permitted thereby. Specifically, with utilization of the side bypass passage 254, as turbine exit velocity increases, this effectively reduces the bypass ratio to actually improve the heating capability thereof. Accordingly, this aspect of the invention takes advantage of the fact that in the heating mode there is a higher turbine exit velocity (therefore m re extreme vel city stratification) which is the

situation in which the reduced bypass rati for improved heating is desired.

Further modifications and adaptations to the internal bypass duct 254 have been discovered to improve th operational characteristics thereof such that the heat exchanger 14 may function not only as a condenser but also as a heating element, i.e. sufficiently sized to impart significant heat transfer to the exhaust flow through duct 13. More particularly, with respect to FIGS. 4-8, the bypass duct 254, in order to work properly, must be wide enough such that ice will not tend to initiate accumulation thereon. That is, the width of bypass duct 254 must be maintained sufficiently wide so as to prevent accumulation of ice that would eventually bridge the width thereof to close the bypass duct. On the other hand, the size of the bypass duct 254 directly relates to the heat transfer characteristics of heat exchanger 14. In other words, if the bypass duct 254 is too large, adequate heat transfer for condensation and water removal and other purposes will not be available short of a very significant increase in size and weight of the overall heat exchanger 14.

To therefore control and minimize the size of heat exchanger 14, the present invention contemplates inclusion of a closure plate 260 secured to the backside of gap 254 (i.e., in alignment with the plane of the exit face 248). Closure plate 260 is rigidly, sealingly, secured to the adjacent sidewalls of the bypass duct 254 so that the exhaust cross-sectional area of duct 254 is substantially reduced. In this manner, the bypass ratio may be reduced substantially, in comparison to the bypass ratio in the absence of closure plate 260, because of the r duced exit fl w area thereof, thereby all wing lower rate of bypass

flow through bypass duct 254 for a given turbine inlet velocity. Thus, while effectively reducing the bypass flow ratio, the width of passage 254 has, very importantly, not been reduced. In this manner, the bypass pressure ratio may be made smaller for a given size and configuration of heat exchanger 14 without increasing the possibility of ice accumulation within the bypass gap. For purposes of definition herein, the rectangular gap 254 is referred to as having a length direction running from face 236 to face 248, a height direction extending vertically in FIGS. 4, 6 and 7, and a width direction extending horizontally in FIGS. 4, 6 and 7.

Also, it has been found to be important that the closure plate 260 be disposed at the rear end of bypass duct 254 rather than more nearly adjacent the inlet face 236. More specifically, at this rearward location closure plate 260 creates a static pressure region 262 immediately in front of plate 260 relative to airflow therethrough. As best depicted in FIG. 8, the bypass air is effectively "pushed around" the static pressure area 262 as shown by the arrows in FIG. 8. In doing so, the very cold bypass air, which may well contain ice particles, avoids impingement directly upon the exposed surface of plate 260 at high velocity. By avoiding impingement of ice particles at high velocity on the plate 260, accumulation of ice thereon is avoided. In other words, by avoiding or reducing impingement of the airflow directly upon surface 260, ice is prevented from forming thereon even th ugh plate 260 is not directly warmed by any heating source.

In this manner, by carefully selecting the length of the closure plate 260, the bypass ratio may be selected to be substantially less than would be created by a gap 254 of equal width but without th closure plate 260. This

dramatically impr ves the design flexibility f r a given size heat exchanger 14. It is believed closure plate 260 can be effective through various lengths thereof from about 30 percent to 70 percent or more of the height of passage 254.

It will be apparent from the foregoing that the very cold bypass airflow passing through bypass duct 254 and exiting out the smaller portion opening thereof at face 248 will promptly come into contact with the aligned end face portion of plenum casing 28 and impinge directly thereon. As noted, such high velocity impingement of very cold air with, perhaps, entrained ice particles is strongly conducive to ice formation on the aligned portion of plenum casing 28. At the same time, design configuration constraints as well as minimization of pressure loss in the airflow dictates that the aligned portion of the plenum casing 28 cannot be remotely disposed from the opening of bypass duct 254. Accordingly, in the present invention the heating manifold 232 is disposed on the exterior surface of plenum casing 28 in alignment with the opening portion of bypass duct 254. The air exhausted from the air bearing system in the air cycle machine 20 is ducted through passage 230 int manifold 232 to create warming on the exterior surface of plenum casing 28 at this point of relatively cold, spot cooling by the bypass flow. In this manner the parasitic flow loss of the air bearing flow is utilized for "sp t" warming of a cold location on the plenum casing 28. In this manner initiation of ice formation at this critical point on the interior of plenum casing 28 is avoided. Normally, the amount of air used for air bearing cooling is betw en 2-5 percent of th system airflow, and is adequate for maintaining the metal temperatur at the cold spot l cation on plenum casing 28 above freezing. Air

manif ld 232 may be readily attached as by welding or mechanical attachment means at the desired cold spot location. Importantly the air manifold 232 has exit openings 233 so as to promote continuous flow of air through the air bearing system.

It will be apparent to those skilled in the art that the closure plate 260 may be utilized in a bypass duct which is located other than at the extreme side as illustrated in the preferred embodiment. That is, the closure plate 260 may be utilized for the same purpose and function in a centrally located bypass duct as illustrated in the Kinsell et al Patent No. 4,352,273. In such instance, the anti-icing features of air bearing exhaust manifold 232 may be equally utilized by moving the chamber 232 into alignment with the bypass duct, wherever it may be.

An alternate arrangement of environmental contr l system 310 is illustrated in FIG. 9. As denoted by common reference numerals the vast majority of elements in system 310 are like in structure and function to the system 10 illustrated in FIG. 1. As described in greater detail below, however, the heat exchange 314 has alternate structure to the heat exchange means 14 illustrated with respect to FIGS. 1-8. Additionally, the system 310 is distinct in the absence of filter 91, and the associated bypass around heat exchanger 16. Further, the arrangement in the system 310 includes an overall bypass passage 340 extending from inlet 11 to outlet 13 and interconnecting with the latter via a schematically illustrated control valve 342. Characteristically the FIG. 9 environmental control system utilizes the airflow passing through the various heat exchangers 42, 48, 16, 314 and the air cycle machine 20 primarily to produce a low temperature

conditioned airfl w through outlet 90 to valve 342. Warm airflow through bypass duct 340 is also presented to valve 342 which then may be modulated to mix the airflows from passages 90 and 340 to produce a desired temperature fl w output in outlet 13.

As such, the heat exchanger 314 acts primarily as a condenser for condensing out of the moisture in the warm airflow passing from passage 82 and plenum casing 330 to plenum casing 332 and to passage 83. Heat exchanger 314 is much smaller than its counterpart in FIG. 1 and is not sized to function in a manner of imparting substantial heating to the very cold sub-freezing airflow presented to casing plenum 326 from the outlet of the turbine 22. As depicted in FIG. 10 the heat exchange 314 does not include a bypass passage, such as the bypass passage 254 of the FIG. 1 embodiment. The FIG. 10 arrangement does however include a backpressure plate 346 disposed closely adjacent to the rear face 348 of the heat exchanger core.

In the FIG. 10 arrangement the very cold sub-freezing airflow from the turbine 22 is presented, again in close-coupled relationship as described above with respect to FIG. 1, into outlet plenum casing 326 for flow thr ugh the core of the heat exchanger from the front surface 336 to rear surface 348 thereof. Flow exiting the rear face of the core passes through the outlet plenum casing 328 for ultimate delivery toward the valve 342 of FIG. 9. With such an arrangement, the backpressure plate 346 acts similarly to that described above with respect t backpressure plate 246 of the FIG. 1-8 embodiment to minimize the flow velocity stratification at the inl t face 336 of the heat exchanger core. By virtue of elimination or substantial reduction in the flow velocity stratificati n across such face, tendencies toward icing

are reduced as described above with respect to FIG. 1. Additionally, the predictability and reliability of the heat exchanger 314 is far better inasmuch as the flow profile is more evenly distributed throughout the entire volume of the heat exchanger core. Thus, the FIG. 9 arrangement which does not utilize a heat exchanger both for condensation and heating functions, but rather only primarily for condensation, may also reap the benefits of the backpressure plate 346 in the same manner as discussed above with respect to FIG. 1.

Another embodiment of the present invention is illustrated in FIG. 11. This is another alternative heat exchange structure 414 which may be utilized as an alternative to the heat exchanger 314 of FIG. 9. The heat exchanger 414 of FIG. 11 does not include a backpressure 346 as depicted above with respect with FIG. 10, but does include a bypass passage 454 which is disposed at the extreme side of the heat exchanger core with only a single one pass heat exchange 456 between the bypass passage 454 and the outer surface 458 of the heat exchanger core. In other respects heat exchanger core 414 has features similar to that illustrated in FIG. 10, i.e. an inlet plenum casing 426 leading to a front face 436 of the heat exchanger core, and a corresponding rear face 448 which leads an outlet plenum casing 428.

In the smaller size heat exchange 414 which, like exchanger 314 of FIG. 10, acts primarily only as a condenser, heat exchanger 414 incorporates the benefits of the side located bypass passage 454. That is, because of the side location of bypass 454 out of alignment with the primary high velocity flow distribution received adjacent the inlet face 436 (as graphically depicted by the conical dashed lines 444 in FIG. 11), the velocity profile at the

inlet to bypass passage 454 remains relatively constant in comparison to that experienced in the center portion of the heat exchange core. As a result, the bypass rati flow through passage 454 can remain relatively predictable and constant. The bypass passage 454 further includes the attendant advantages discussed with respect to FIGS. 1-8 of avoidance of ice particle impingement.

Claims:

1. A heat exchanger for conditioning a working fluid, comprising:

a heat exchange core having first and second passageways for heat exchange between the fluid carried therein;

first inlet and outlet plenum chambers at opposite inlet and outlet faces of said core, said first inlet and outlet plenum chambers communicating with said first passageway, said first inlet plenum chamber receiving and arranged such that the subfreezing working fluid impinges on said inlet face of the core with a stratification of flow velocities across said inlet face, the highest flow velocities being concentrated in a centered portion of said inlet face, said first outlet plenum chamber receiving and collecting the working fluid discharged from said first passageway through said outlet face for delivery to a point of use external to said heat exchanger;

second inlet and outlet plenum chambers communicating with said second passageway for carrying relatively warm working fluid flow through the heat exchange core in heat exchange relationship with colder fluid flow in said first passageway; and

flow redistributing means disposed in said first outlet plenum chamber for minimizing said stratification of flow velocities at said inlet face of the core.

2. A system as set forth in Claim 1, wherein said flow redistributing means comprises a back pressure barrier disposed closely adjacent to but spaced from said outlet face.

3. A system as set forth in Claim 2, wherein said barrier covers at least 30 percent of said outlet face.

4. A system as set forth in Claim 3, wherein said barrier covers between 30 percent and 80 percent of said outlet face.

5. A system as set forth in Claim 4, wherein said barrier covers about 60 percent of said outlet face.

6. A system as set forth in Claim 2, wherein said barrier comprises a thin, solid plate affixed to said core, and spaced slightly from said outlet face.

7. A system as set forth in Claim 6, wherein said plate is aligned with that portion of the inlet face wherein the highest flow velocities of said subfreezing fluid flow is concentrated.

8. A system as set forth in Claim 7, wherein said portion of the inlet face is the center portion thereof.

9. A heat exchanger for conditioning a working fluid, comprising:

a heat exchange core having first and s cond passageways for heat exchange between the fluid carri d therein;

first inlet and outlet plenum chambers at opposite inlet and outlet faces of said core, said first inlet and outlet plenum chambers communicating with said first passageway, said first inlet plenum chamber receiving working fluid in a subfreezing condition and arranged such that the subfreezing working fluid impinges on said inlet face of the core with substantially varying, stratified flow velocities across said inlet face, the highest flow velocities being concentrated in a c ntered position f said inlet face, said first outlet plenum chamber r ceiving and collecting the working fluid discharged from

said first passageway through said outlet face for delivery to a point of use external to said heat exchanger;

second inlet and outlet plenum chambers communicating with said second passageway for carrying relatively warm working fluid flow through the heat exchange core in heat exchange relationship with colder fluid flow in said first passageway; and

a bypass passage contained within the confines of said heat exchange core and extending from said inlet fac to said outlet face in parallel flow relationship to said first passageway, said bypass passage located at said inlet face to be exposed only to relative low flow velocities of said stratified velocities of subfreezing working fluid impinging thereon.

10. A system as set forth in Claim 9, wherein said bypass passage is disposed at an extreme side of said inlet face, said second passageway including subpassages on both sides of said bypass passage.

11. A system as set forth in Claim 10, wherein said bypass passage is of generally rectangular configurati n in the planes of said inlet face and said outlet face with a width substantially less than its height, said rectangular bypass passage having a length extending fr m said inlet face to said outlet face.

12. A system as set forth in Claim 11, further including a closure plate sealingly affixed within said bypass passage at a location adjacent said outlet face, said closure face extending completely across the width of said bypass passage, and extending a preselected distance along the height of said bypass passage.

13. A system as set forth in Claim 12, wherein said preselected distance corresponds to a desired ratio of fluid flow through said bypass passage to fluid flow through said first passageway.

14. A system as set forth in Claim 12, wherein said preselected distance is between 30 percent and 70 percent of the full height of said bypass passage.

15. A heat exchanger for conditi ning a working fluid, comprising:

a heat exchange core having first and second passageways for heat exchange between the fluid carried therein;

first inlet and outlet plenum chambers at opposite inlet and outlet faces of said core, said first inlet and outlet plenum chambers communicating with said first passageway, said first inlet plenum chamber receiving working fluid in a subfreezing condition and arranged such that the subfreezing working fluid impinges on said inl t face of the core with substantially varying fl w velocities across said inlet face, the highest fl w velocities being concentrated in a centered position of said inlet face, said first outlet plenum chamber receiving and collecting the working fluid discharged fr m said first passageway through said outlet face f r delivery to a point of use external to said heat exchanger;

second inlet and outlet plenum chambers communicating with said second passageway for carrying relatively warm working fluid flow through the heat exchange core in heat exchange relationship with colder fluid flow in said first passageway;

a bypass passage defined within the confines of said core and extending from said inlet face to said outlet face for carrying working fluid from said first inlet plenum chamber to said first outlet plenum chamber in parallel, bypassing relationship to said first passageway; and

a closure plate sealingly affixed within said bypass passage at a location adjacent said outlet face to extend partially across said bypass passage in a direction reducing bypass flow ther through without reducing the minimum dimension f said bypass passage.

16. A system for conditi ning a working fluid to be distributed from a source thereof to a point of use, comprising:

an air cycle machine including a compressor mechanically driven by a turbine, said compressor receiving working fluid from said source and operable to compress and heat the working fluid, said turbine operable to expand and cool the working fluid to subfreezing conditions prior to its delivery to the point of use;

first means for receiving and cooling the working fluid discharged from said compressor;

reheater means for receiving and further cooling the working fluid discharged from said first means;

condenser heat exchange means, receiving working fluid discharged from said reheater means, for condensing vapor entrained in the working fluid, said condenser means receiving the subfreezing working fluid discharged from said turbine to effect said condensing;

separator means, receiving the working fluid and condensed vapor discharged from said condenser means, for separating and removing the condensed vapor from the working fluid, at least a portion of the working fluid discharged from said separator means being direct d through said reheater means to effect said further cooling of the working fluid discharged from said first means, and substantially all of the working fluid discharged from the separator means being delivered to said turbine;

said condenser heat exchange means defining a core having a first and second passageways arranged for heat exchange between the fluids therein, said first and second passageways being configured and sized to heat the subfreezing working fluid discharged from said turbine, said condenser heat exchanger m ans including:

first inlet and outlet plenum chambers at opposite inlet and outlet faces of said core, said first inlet and outlet plenum chambers communicating with said first passageway, said first inlet plenum chamber receiving the subfreezing working fluid discharged from said turbine and being sufficiently close-coupled to said turbine to produce a substantial stratification of flow velocity of the subfreezing working fluid impinging upon said inlet face of the core, said first outlet plenum chamber receiving and collecting the working fluid discharged from the first passageway for delivery thereof t the point of use;

second inlet and outlet plenum chambers communicating with said second passageway, said second inlet plenum chamber receiving the working fluid discharged from said reheater means, and said second outlet plenum chamber receiving and collecting the working fluid discharged from said second passageway for delivery thereof to said separator means; and

flow redistributing means disposed in said first outlet plenum chamber for minimizing said stratification of flow velocities at said inlet face of the core.

17. A system as set forth in Claim 16, wherein said condenser heat exchange means further includes a bypass passage contained within the confines of said condenser heat exchange means and extending from said inlet face to said outlet face in parallel flow relatively to said first passageway, said bypass passage located at said inlet face to be exposed only to relative low flow velocities of said stratified velocities of subfreezing working fluid impinging thereon.

18. A system as set forth in Claim 17 wherein said air cycle machine includes a foil journal air bearing and a waste flow exhaust passage for carrying heated, exhaust airflow from said foil bearing, said first outlet plenum chamber having a heating manifold carried on an external surface thereof in general alignment with fluid flow exiting said bypass passage, said heating manifold receiving heating airflow from said waste flow exhaust passage.

19. A method for conditioning working fluid comprising the steps of:

cooling a hot, pressurized flow of working fluid;

compressing the cooled working fluid;

further cooling the compressed working fluid to remove the heat of compression introduced therein;

condensing water vapor entrained in the further cooled working fluid;

removing the condensed water vapor to produce a drier working fluid;

expanding the dried working fluid across a turbine to produce a subfreezing working fluid flow;

directing the subfreezing working fluid flow onto the inlet face of a heat exchange core with substantially varying flow velocities across the inlet face;

varying the working fluid flow velocity being directed toward the inlet face to create yet great r variations in flow velocities across said inlet face; and

redistributing the flow exiting the heat exchange core to minimize the variation in flow velocities at the inlet face thereof.

20. A method as set forth in Claim 19, including the step of bypassing a portion of the flow around the heat exchange core the rough a bypass passage contained therewithin, the bypass passage opening onto the inlet face at a side location thereon wherein flow velocity variation is minimized.

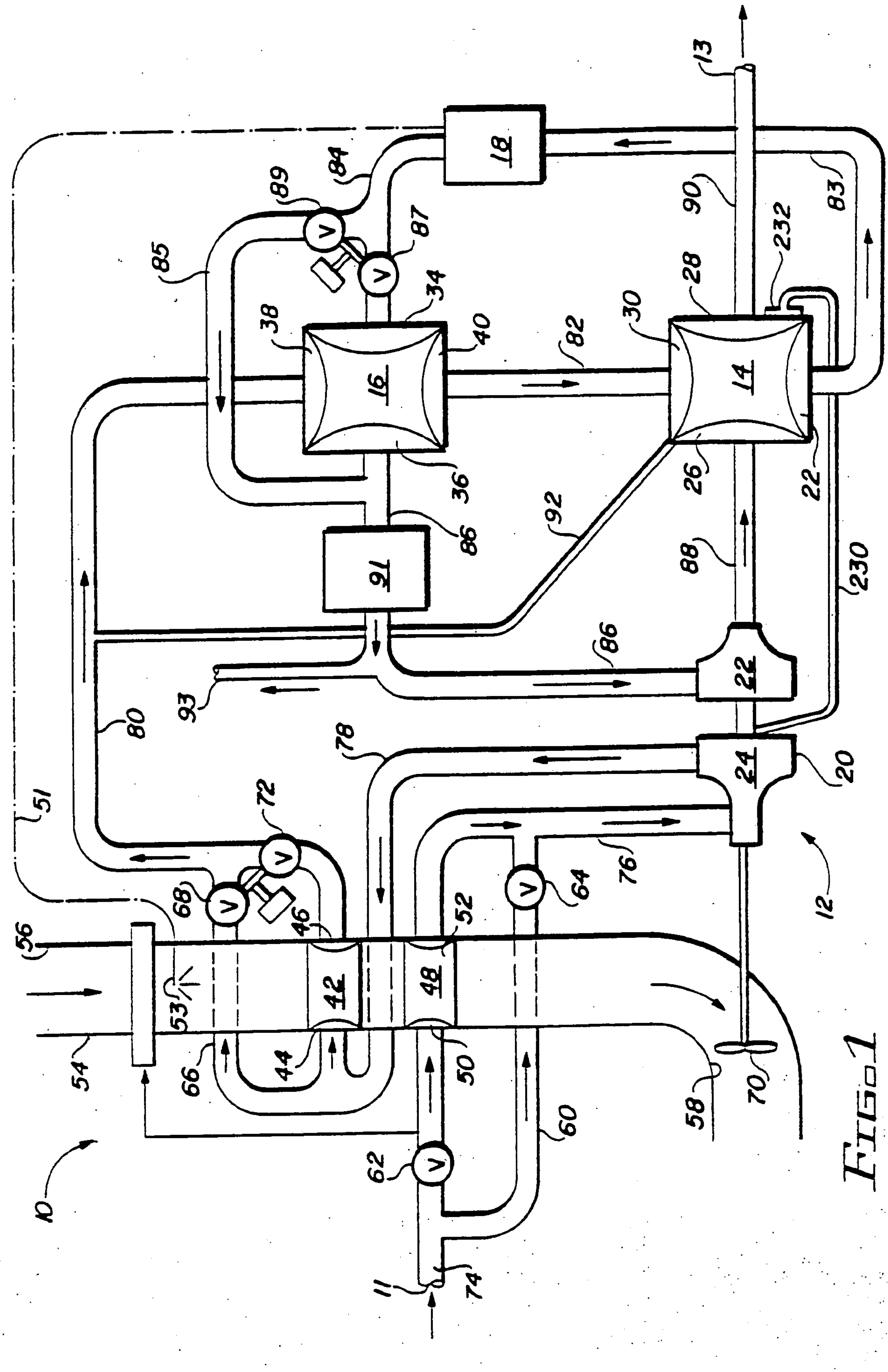

FIG. 1

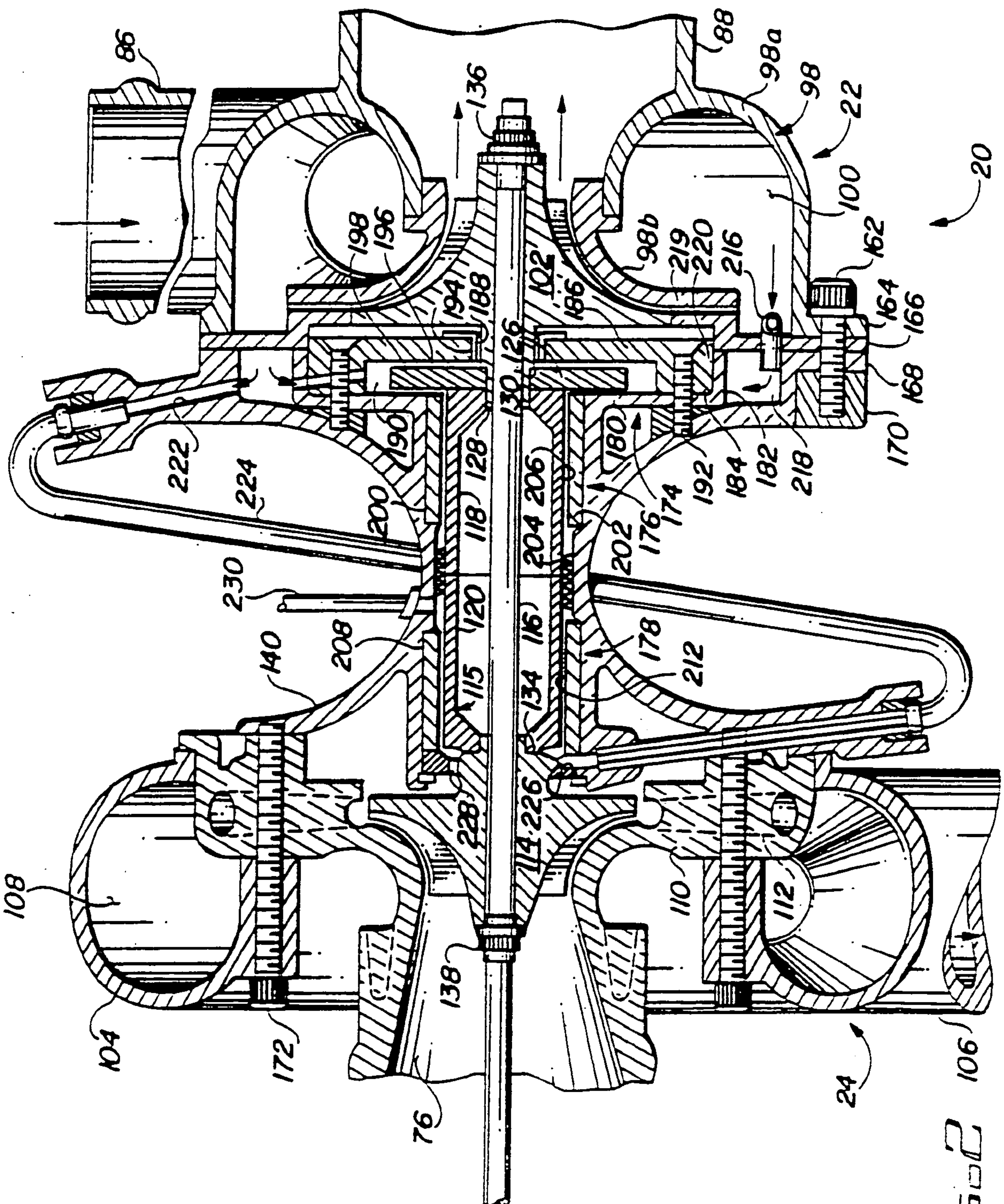
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FIG. 2

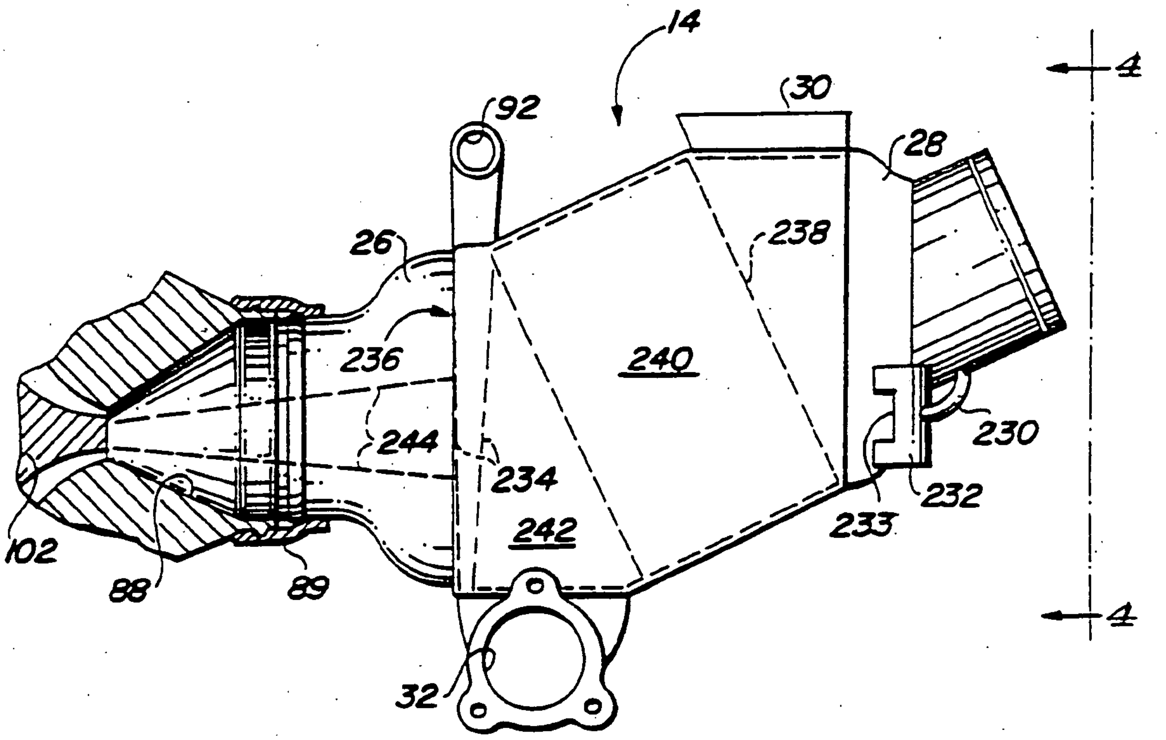
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FIG. 3

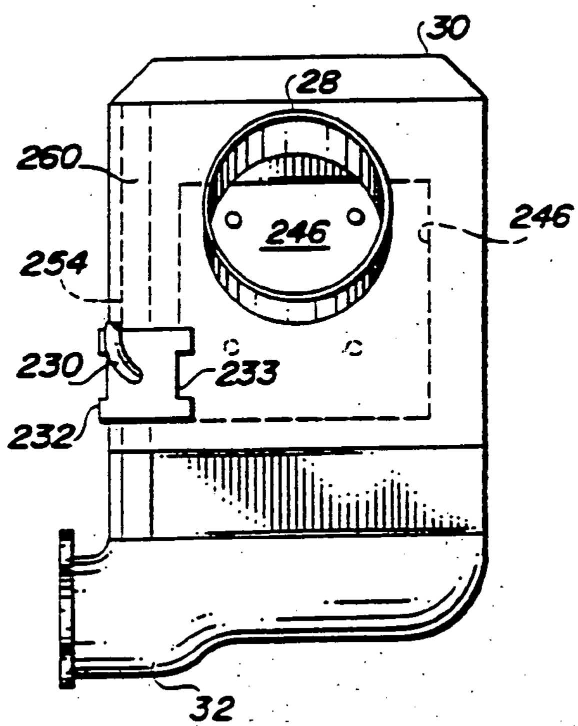
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FIG. 4

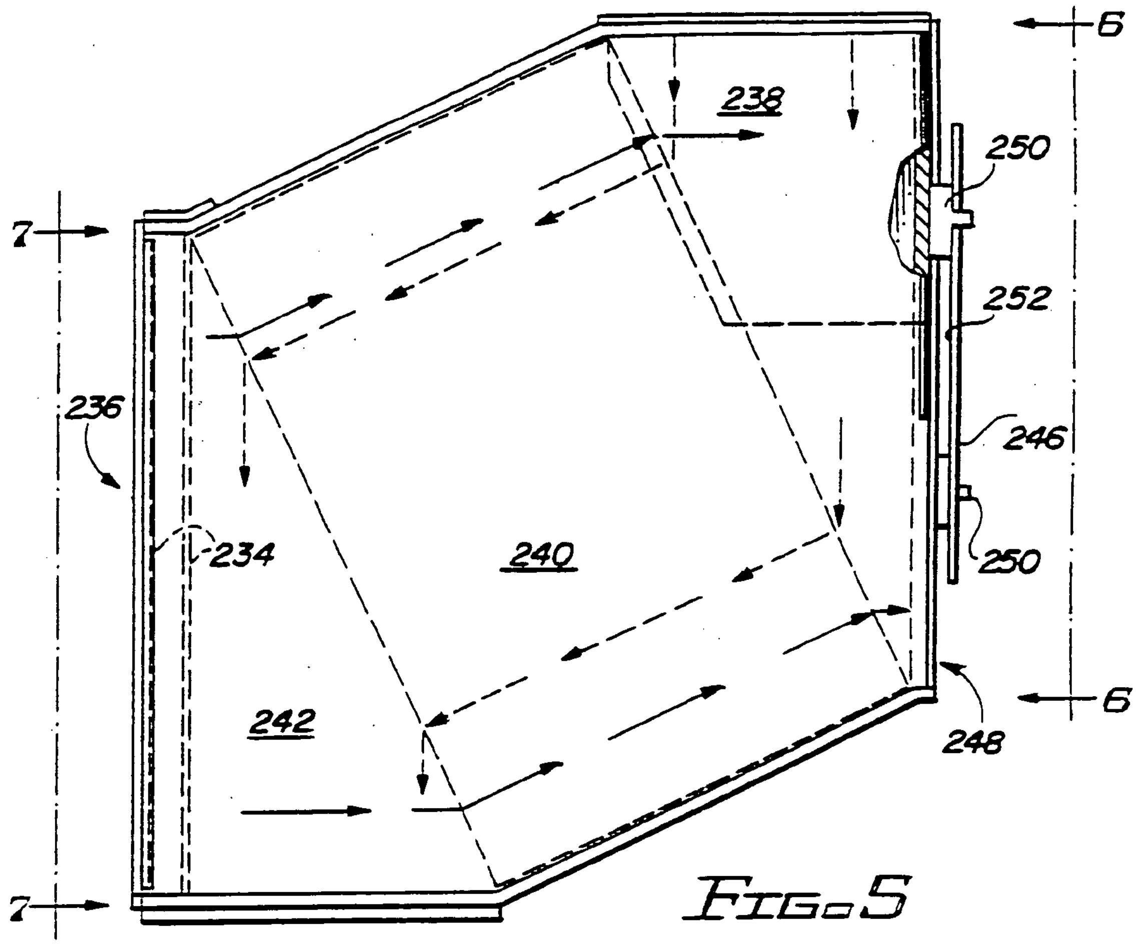
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FIG. 5

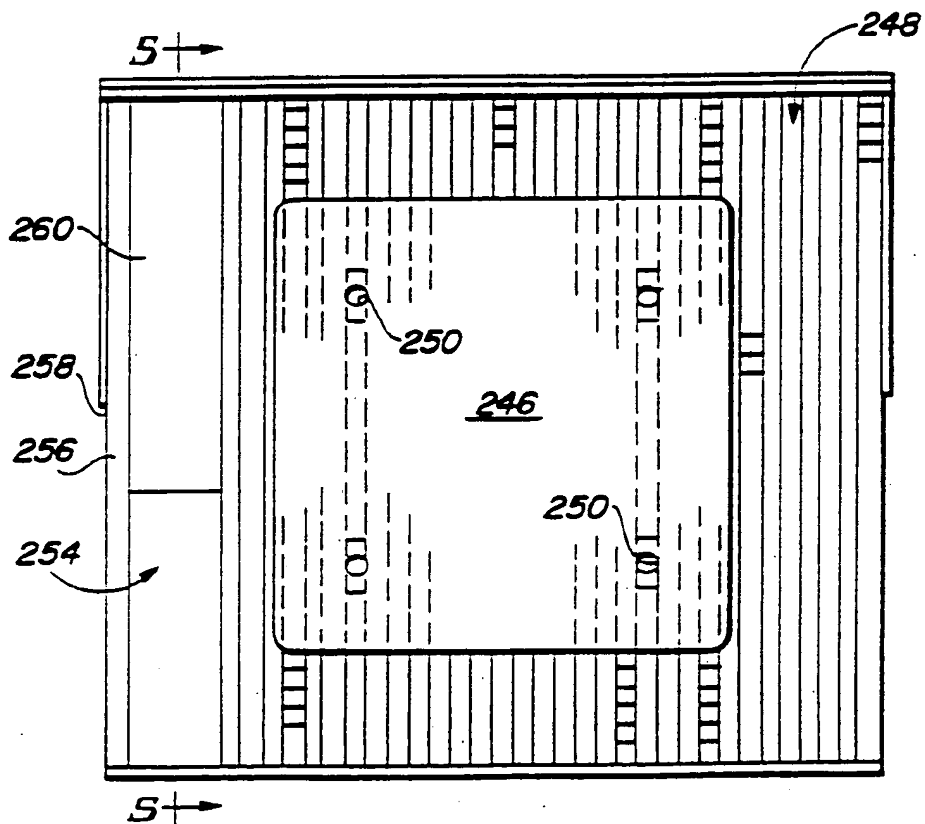
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FIG. 6

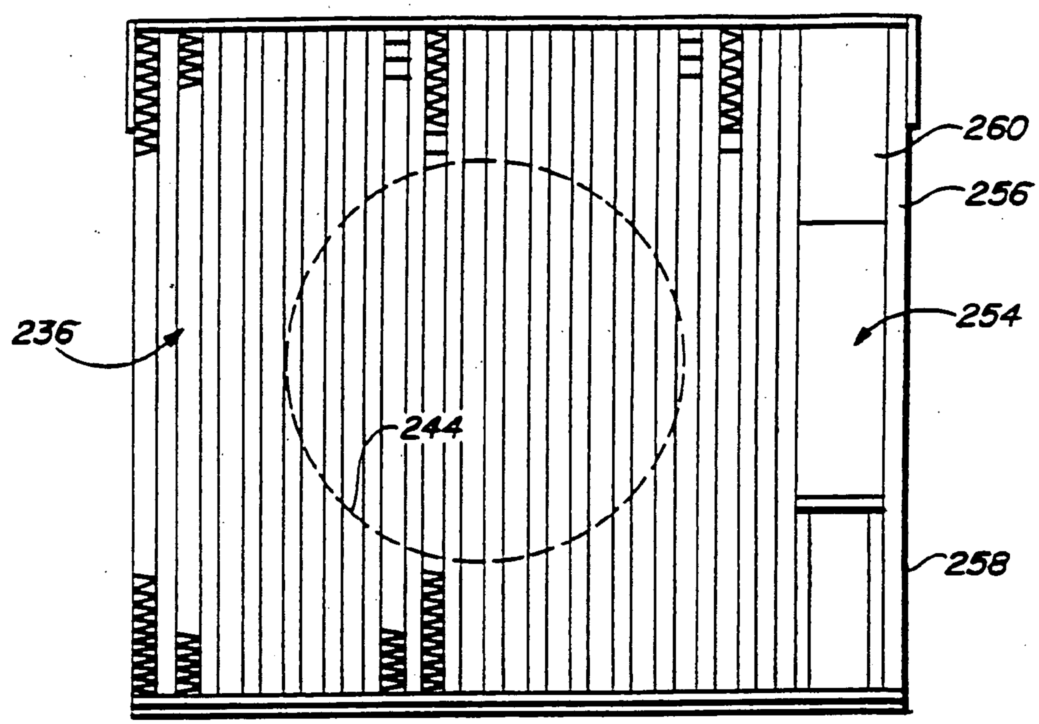

FIG. 7

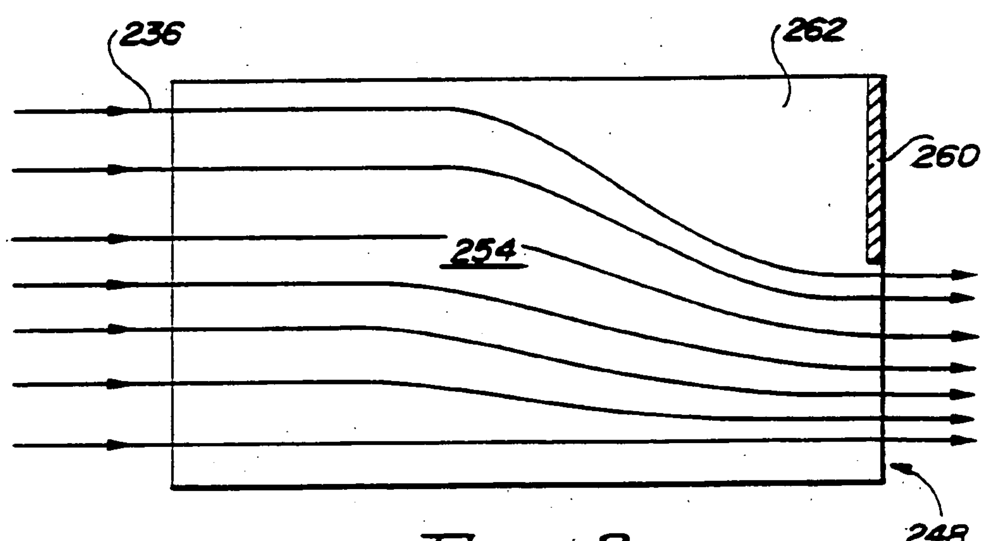

FIG. 8

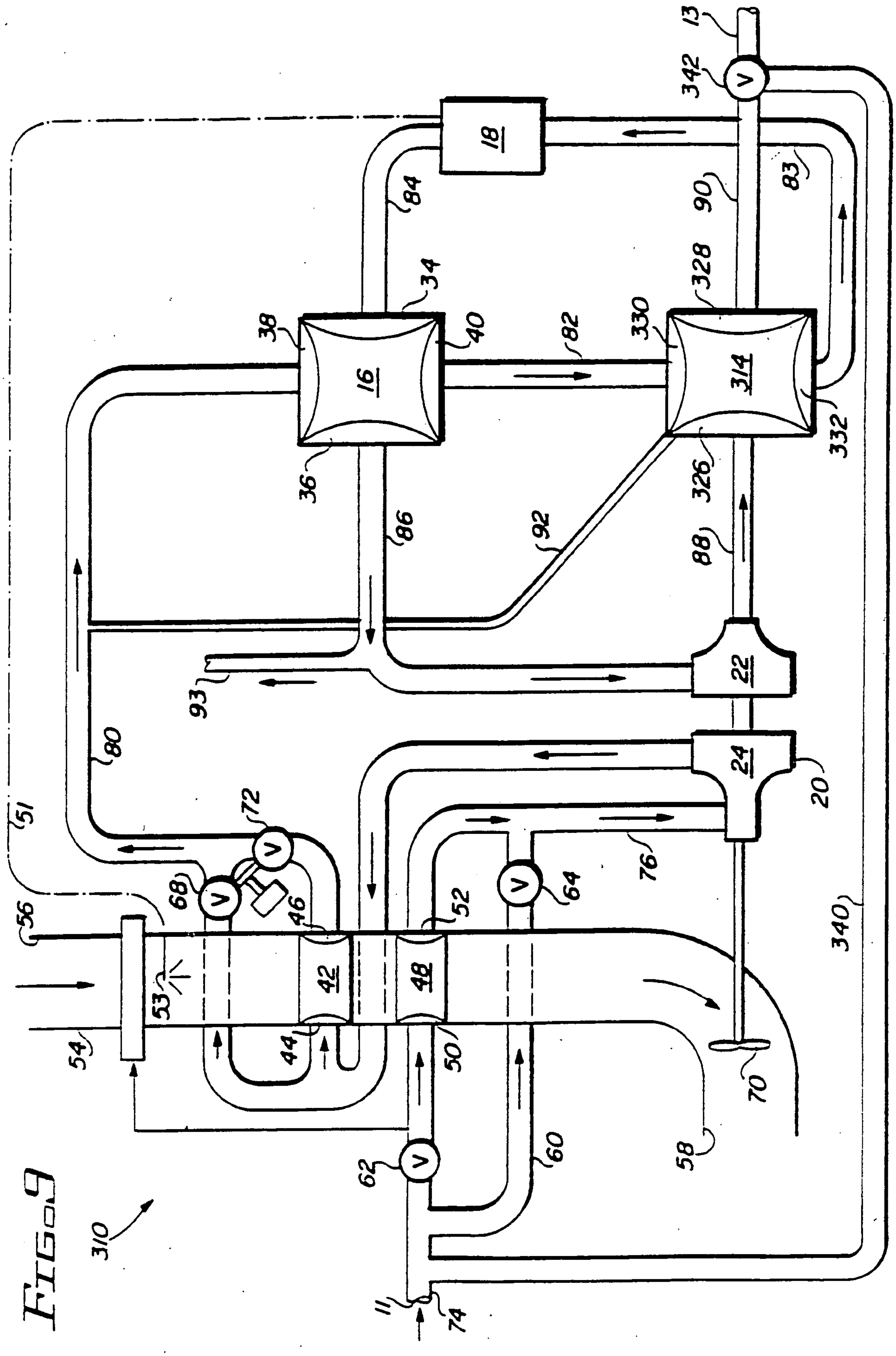
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FIG. 9

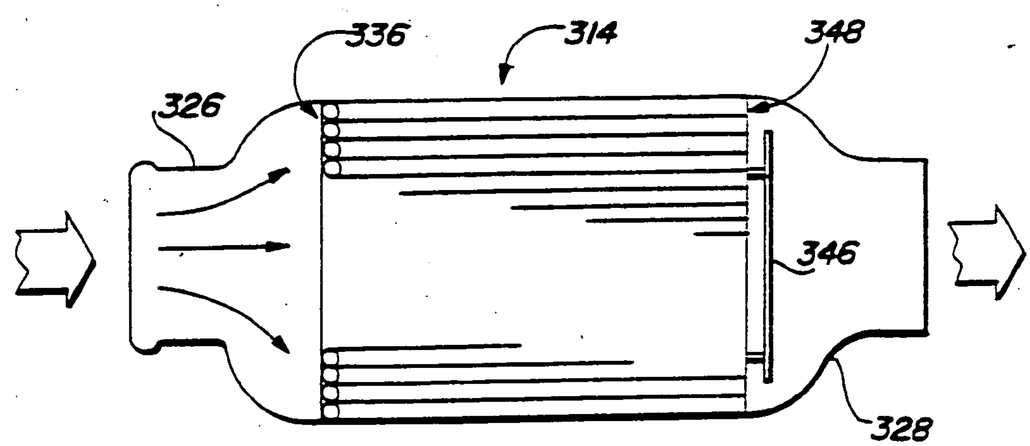

FIG. 10

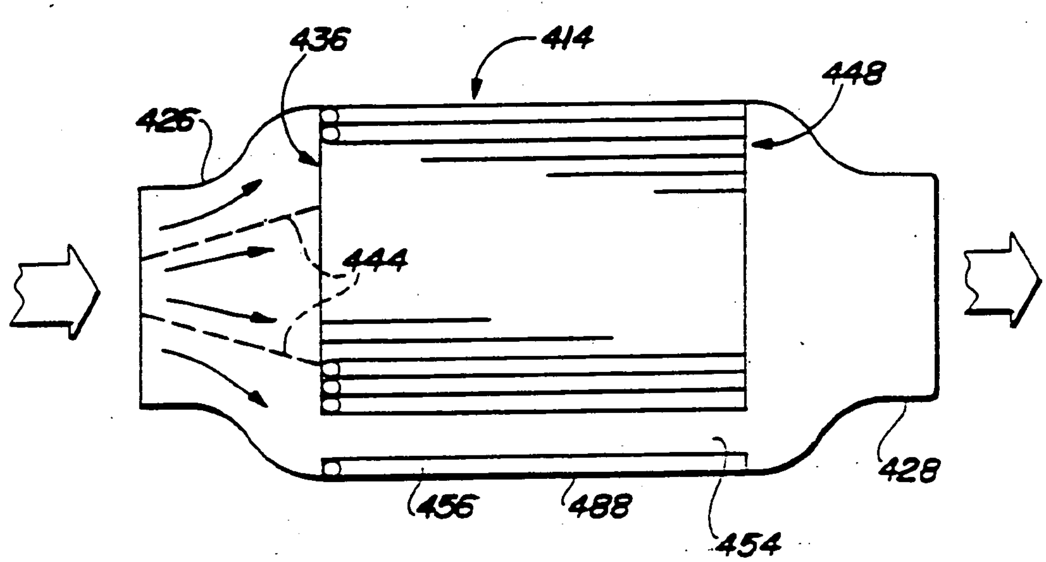

FIG. 11

INTERNATIONAL SEARCH REPORT

International Application No PCT/US 91/01044

I. CLASSIFICATION OF SUBJECT MATTER (If several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5 B64D13/08

II. FIELDS SEARCHED

Minimum Documentation Searched[7]

Classification System	Classification Symbols
Int.Cl. 5	B64D ; F28F ; F25B ; F24F

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT[9]

Category *	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No.[13]
A	US,A,4352273 (KINSELL ET AL.) 05 October 1982 see column 3, line 42 - column 5, line 57 see column 5, line 67 - column 6, line 28 see column 6, line 58 - column 7, line 40; figures 1-3 (cited in the application) ---	1, 9, 15, 16, 19
A	US,A,4580406 (NIMS, R.) 08 April 1986 see column 1, lines 59 - 64 see column 2, lines 13 - 41; claim 1; figure 1 ---	1, 9, 15, 16, 19
A	EP,A,341663 (UNITED TECHNOLOGIES CO.) 15 November 1989 see page 2, column 2, lines 14 - 39 see page 3, column 3, lines 9 - 30 see page 4, column 6, lines 18 - 40; figures 2, 5 ---	1, 15, 16, 19

-/--

* Special categories of cited documents : [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
31 JULY 1991	23. 08. 91
International Searching Authority	**Signature of Authorized Officer**
EUROPEAN PATENT OFFICE	RIVERO C.G.

III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)

Category *	Citation of Document, with indication, where appropriate, of the relevant passages	Relevant to Claim No.
A	US,A,4198830 (CAMPBELL) 22 April 1980 see column 2, lines 9 - 40 see column 2, line 66 - column 4, line 5; figure (cited in the application) ---	1, 9, 15, 16, 19

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO.

PCT/US 91/0102
SA 46523

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information. 31/07/9

Patent document cited in search report	Publication date	Patent family member(s)		Publication date
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		JP-C-	1305616	13-03-86
		JP-A-	55165489	23-12-80
		JP-B-	60027915	02-07-85
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		JP-A-	61139597	26-06-86
EP-A-341663	15-11-89	US-A-	4862952	05-09-89
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